Chapter

Modeling Friction Losses in the Water-Assisted Pipeline Transportation of Heavy Oil

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Abstract

In the lubricated pipe flow (LPF) of heavy oils, a water annulus acts as a lubricant and separates the viscous oil from the pipe wall. The steady state position of the annular water layer is in the high shear region. Significantly, lower pumping energy input is required than if the viscous oil was transported alone. An important challenge to the general application of LPF technology is the lack of a reliable model to predict frictional pressure losses. Although a number of models have been proposed to date, most of these models are highly system specific. Developing a reliable model to predict pressure losses in LPF is an open challenge to the research community. The current chapter introduces the concept of water lubrication in transporting heavy oils and discusses the methodologies available for modeling the pressure drops. It also includes brief descriptions of most important pressure loss models, their limitations, and the scope of future works.

Keywords: lubricated pipe flow, continuous water-assisted flow, core annular flow, CFD, friction factor, data analysis

1. Introduction

1.1 Background

The reserve of nonconventional heavy oils is one of the most important petroleum resources in the current world [1, 2]. These oils are highly asphaltic, dense, and viscous compared to conventional oils, such as Brent and West Texas Intermediate [3, 4]. The density is comparable to that of water, and the viscosity can be greater than that of water by more than five orders of magnitude at room temperature [5, 6]. This type of highly viscous oils is produced using a variety of mining and *in situ* techniques [7]. After extraction, the oil is delivered from the production site to a central processing/upgrading facility. A number of pipeline transportation methods are available for the transportation. The conventional transporting technologies involve viscosity reduction through heating or dilution [1, 3, 4, 8].

The focus of the current chapter is the lubricated pipe flow (LPF) of heavy oils, where a water annulus separates the viscous oil-core from the pipe wall. It is an alternative flow technology, which is more economic and environmentally friendly than conventional heavy oil transportation technologies [9, 10]. The benefit of LPF is that it is a specific flow regime in which a continuous layer of water can be found near the pipe wall. As wall shear stresses are balanced by pressure losses in any kind of pipeline transportation, this flow system requires significantly less pumping energy than would be required to transport the viscous oil alone at comparable process conditions [10, 11–17].

A number of industrial scale applications of LPF are reported in the literature. For example, a 38.9-km long lubricated pipeline having 6 inch diameter was successfully operated by Shell for more than 12 years in California [18]. The frictional pressure loss for this pipeline was not only orders of magnitude less than that for transporting heavy oil but also quite comparable to the loss for transporting water [6]. The pipeline was operated by adding up to 30 vol% water. A number of water lubricated pipelines were used to transport heavy oil at Lake Maracaibo in Venezuela [1]. One of the challenges the operators faced to run these pipelines was cumulative wall fouling. Different operational measures, such as increasing water fraction or water flow rate and changing the water composition were taken to control the fouling. However, these measures were never sufficient to stop wall fouling. Water lubricated pipe flow technology was also used in Spain for the purpose of transporting heavy fuel oil [6]. Syncrude Canada Ltd. transported bitumen froth (a mixture of 60% bitumen, 30% water, and 10% solids) from a remote extraction plant to upgrading facility; they used a 35-km long and 36-inch diameter lubricated pipeline [12, 19, 20]. The lubrication process in the Syncrude pipeline produced a fouling layer of oil on the pipe wall. The thickness of the fouling layer was approximately 5% of the pipe's internal diameter [12, 19]. At present, Brazilian oil producers are in the process of producing viscous oils from off-shore reservoirs with the application of water lubricated flow in vertical pipelines [21, 22].

A concerning phenomenon during the lubricated pipe flow of viscous heavy oil or bitumen is wall fouling [1, 3]. The probable LPF regime is presented in **Figure 1**. A wall fouling layer of oil is shown to surround a water annulus lubricating the viscous oil core. Although a number of experimental studies demonstrated the fouling layer to be a natural and inevitable consequence of the lubrication process, the mechanism of wall fouling in LPF has not been studied in detail [7, 12, 17, 19]. The application of LPF where the phenomenon of wall fouling must be accepted under regular operating conditions is sometimes referred to as "continuous water assisted flow (CWAF)" [13].

1.2 Lubricated pipe flow

Successful operation of a water lubricated pipeline is dependent on a few critical flow conditions. The preliminary requirement for establishing LPF is the simultaneous pumping of heavy oil and water in the pipeline. This kind of pumping into a horizontal pipeline can result in different flow regimes, depending upon the superficial velocities and the properties of oil [18, 24, 25]. The prominent flow regimes

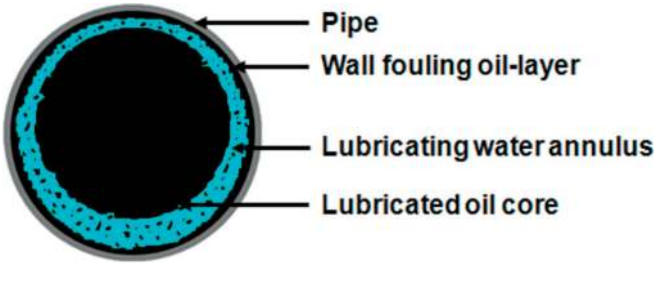

Figure 1.
Hypothetical presentation of the flow regime in a water lubricated pipeline [23].

are dispersed, stratified flow, bubbles, slugs, and lubricated flows. The boundaries between the flow regimes are not well defined [7, 18]. It is possible to describe qualitatively the transition from a flow regime to the other one on the basis of similar regime transitions in gas-liquid flow systems [13]. At lower flow rates of the fluids, stratified flow can be expected [26, 27]. The relative positions of the oil and water in this kind of flow regime are controlled by the effect of gravity, that is, the difference between the liquid densities. If the density of water is higher than that of oil, oil is to float on water and vice versa. By increasing the water flow rate, the stratified flow regime may be transformed into bubble or slug flow. The increased flow rate is likely to increase the kinetic energy and turbulence of the water, which results in waves at the oil-water interface and, ultimately, transforms the stratified oil into bubbles or slugs. Further increase in the water flow rate may split bubbles or slugs into smaller droplets of oil. Contrariwise, increasing the oil flow rate at a constant water flow can promote coalescence of bubbles or slugs, which may produce the water lubricated flow regime [24, 25].

The minimum velocity for the mixture of heavy oil and water required to obtain the water lubricated flow regime in a horizontal pipeline has been reported as 0.1–0.5 m/s for different applications [10, 12, 13, 15, 28]. In addition to the minimum velocity criterion, sustainable lubricated pipe flow also requires a minimum water fraction, typically between 10 and 30% [1]. A greater percentage of lubricating water does not cause a significant reduction in the pressure loss; even if it reduces the pressure loss to some extent, it also reduces the amount of oil transported per unit of energy consumed [10, 13, 20]. Water lubrication is usually identified from pressure loss measurements [13]. The establishment of lubricated pipe flow is typically associated with a significant and nearly instantaneous reduction in frictional pressure losses [20].

As mentioned earlier, a significant concern during the application of lubricated pipe flow is that a minor fraction of the transported oil tends to adhere to the pipe wall, which eventually leads to the formation of an oil layer on the pipe wall [1, 3, 12, 13, 15, 18, 19, 29]. Frictional pressure losses in a "fouled" pipe, that is, with an oil coating on the wall, are higher compared to those for transportation of the same mixture in an unfouled pipe [15, 30]. Nevertheless, the frictional losses with wall fouling are substantially lower than that would be expected for transporting only heavy oil [10, 20, 29].

Wall fouling is practically unavoidable in the water lubricated pipeline transportation of viscous oils [10, 12, 13, 15, 20]. Varying degrees of wall fouling are experienced in the applications of this pipe-flow technology. Different descriptions have been used in the literature to classify these applications, for example:

a. Core annular flow [11, 30]

b. Self-lubricated flow [12]

c. Continuous water assisted flow [10, 13]

Lubricated pipe flow has been used in this chapter to refer to any of these flow types, despite the fact that they exhibit quite different characteristics.

Core annular flow (CAF) primarily denotes an idealized version of lubricated pipe flow. It involves a core of viscous oil lubricated by a water annulus through a pipe with a clean (unfouled) wall [11, 29, 31]. Many research studies published in the 1980s and 1990s focused exclusively on CAF, for example, [11, 31, 32]. In most of these studies, wall fouling was either minimized or avoided through prudent selection of operating conditions, such as water cut and construction material of pipe. In pilot-scale and industrial operations, attempts to operate CAF pipeline usually required expensive mitigation strategies to handle wall fouling. In most published cases, it was impossible to avoid wall fouling (see, for example [15, 33]).

The self-lubricated flow (SLF) and continuous water assisted flow (CWAF) are the commonly applied forms of LPF in the industry. As mentioned earlier, the SLF refers to the water lubricated pipeline transportation of a viscous mixture known as bitumen froth containing approximately 60% bitumen, 30% water, and 10% solids by volume [12, 19, 20]. The water fraction in the froth lubricates the flow; additional water is usually not added. In a SLF pipeline, water assist appears to be intermittent, and the oil core may touch the pipe wall at times [10, 12, 29]. Continuous water assisted flow denotes the pipeline transportation of heavy oil or bitumen when the water lubrication is more stable and the oil core touches the pipe wall infrequently [10, 13, 34]. Approximately 20–30 vol% water required to produce lubricated flow is supplier from an external source to a CWAF pipeline. Both SLF and CWAF involve wall fouling. For example, the thickness of fouling layer was measured from 5.5 to 8.5 mm in a 150-mm SLF pipeline transporting bitumen froth at 25°C [19]. Similar thicknesses in a 100-mm CWAF pipeline were found to vary from 1 to 5 mm depending on the operating temperature and mixture velocity [10, 23].

1.3 Modeling LPF pressure losses

Lubricated pipe flow has been applied in a specific industrial context for transporting viscous oils like heavy oil and bitumen with limited success in many cases [1, 3, 9, 18, 20, 21]. A challenge to the broader application of LPF technology is the lack of a reliable model to predict frictional pressure losses, even though numerous empirical (e.g., [12, 13]), semi-mechanistic or phenomenological (e.g., [10, 11, 15]) and idealized models (e.g., [14, 32, 33, 35–37]) have been proposed to date. The existing models were developed based on either single-fluid or two-fluid approach. A critical analysis of these models is important to underscore their limitations and to realize the scope of developing new approach to model LPF frictional losses.

1.3.1 Single-fluid approach

Single-fluid models are also known as equivalent fluid models. This kind of models generally takes an engineering approach to predict the pressure gradients. The flow system is modeled by considering the flow of a hypothetical fluid under comparable LPF process conditions. In some cases, this hypothetical fluid is water [10, 12, 13, 15]. In other cases, the properties of this fluid are determined using the mixture properties [11]. The flow regime in a single-fluid model is assumed to be in turbulent state, and the friction factor is recognized as inversely proportional to the n^{th} power of a representative Reynolds number (Re), that is, $f = K/Re^n$. The constants K and n are either determined empirically or simply assigned. The Reynolds number is defined with respect to the properties of the hypothetical liquid and the pipeline conditions: an equivalent density (ρ) and viscosity (μ) of the hypothetical liquid, the pipe diameter (D), and the average mixture velocity (V). The famous Blasius formula ($f = 0.079/Re^{0.25}$) is often the basis of single-fluid modeling approach. This empirical law was originally proposed for the turbulent flow of water in a smooth pipe. The value of K in Blasius formula ($K = 0.079$) can be tweaked to take into account the equivalent hydrodynamic roughness produced by the pipe-wall and/or wall fouling layer. Thus, single-fluid models take an empirical approach to predict pressure loss for lubricated pipe flow; the actual physical mechanisms governing pressure losses in a water lubricated pipeline are mostly disregarded.

1.3.2 Two-fluid approach

There are a few two-fluid models available in the literature [23, 32, 33, 38]. However, most of these were proposed for smooth pipe CAF, that is, this kind of models does not take into account the hydrodynamic roughness. As a result, these models are not suitable for SLF or CWAF. However, these models do have an advantage over single-fluid models. The actual mechanism of frictional pressure loss is addressed to some extent while developing a two-fluid model. The modeling approach is described in details with two examples as follows.

Oliemans et al. [32] described the mechanism of frictional losses in their pioneering model developed for a CAF system. They identified the shear in the turbulent water annulus as the major contributing factor to pressure losses. However, they had to empirically address two important aspects of core annular flow: physical roughness on the oil core and water holdup. They also used a couple of idealized concepts like Reynold's lubrication theory and Prandtl's mixing length. This two-fluid model systematically underpredicted the CAF pressure losses. Also, the implementation of the model is not straightforward.

Ho and Li [31] adapted the key features of Shi et al. [33] to develop another two-fluid model. They recognized the major source of frictional pressure loss in CAF to be the shear in the turbulent water annulus and modeled the turbulence based on the concept of Prandtl's mixing length. They also considered the oil core to be a plug having a rough surface. However, instead of empirically quantifying this roughness like [33], the complexity of physical roughness was simplified in [32] based on the concept of hydrodynamic roughness. An idealized core annular flow regime was subdivided into four hypothetical zones as presented in **Figure 2**, which also depicts the dimensionless distances of these zones from the stationary pipe wall. The velocity profiles in the sublayers are usually presented using these nondimensional terms. The relationships of flow rate and pressure drops were obtained by integrating these velocity profiles with respect to the dimensionless distance. The equations are presented in **Table 1**.

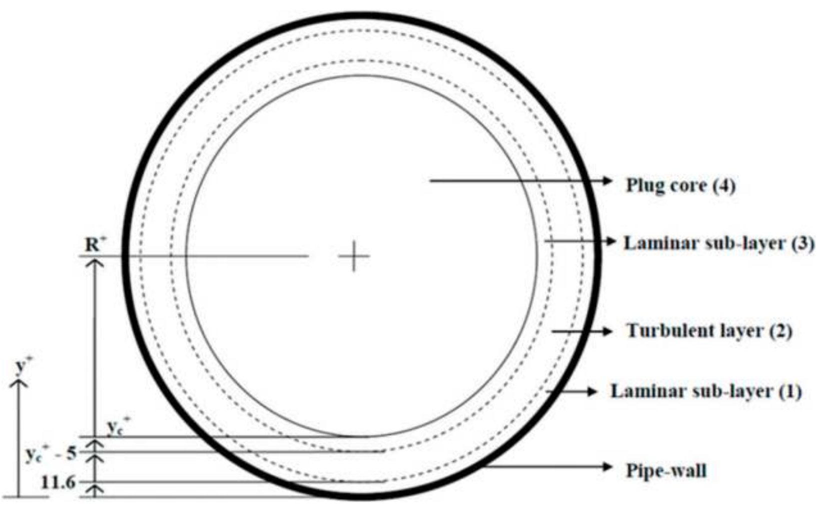

Figure 2.
Hypothetical subdivision of perfect or ideal core annular flow into four zones showing dimensionless distances from the pipe wall [31].

Zone (Figure 2)	Equations	Range
Laminar sublayer (1)	$u_1^+ = y^+$	$0 \le y^+ \le 11.6$
Turbulent layer (2)	$u_2^+ = 2.5\ln(y^+) + 5.5$	$11.6 \le y^+ \le y_c^+ - 5$
Laminar sublayer (3)	$u_3^+ = 2.5\ln(y_c^+ - 5) - y_c^+ + 10.5 + y^+$	$y_c^+ - 5 \le y^+ \le y_c^+$
Plug core (4)	$u_4^+ = 2.5\ln(y_c^+ - 5) + 10.5$	$y_c^+ \le y^+ \le R^+$
(1) + (2) + (3)	$Q_w = 2\pi(\nu_w^2/v^*)[(2.5R^+y_c^+ - 1.25y_c^{+2})\ln(y_c^+ - 5) + 3R^+y_c^+ - 2.125y_c^{+2} - 13.6R^+]$	$0 \le y^+ \le y_c^+$
(4)	$Q_o = \pi(\nu_w^2/v^*)(R^+ - y_c^+)^2[2.5(\ln y_c^+ - 5) + 10.5]$	$y_c^+ \le y^+ \le R^+$

Table 1.
Velocity profiles and equations relating flow rates and pressure losses [31].

The principal focus of Ho and Li [31] was the water annulus in a CAF pipeline. The annular thickness was the most important parameter in the two-fluid model. However, they had to determine this thickness empirically. Moreover, they used the idealized concept of perfect CAF, that is, the perfectly concentric orientation of the oil core in the pipe, even though the orientation is more likely to be eccentric [29, 33]. The eccentricity of the oil core has a consequential effect on the CAF pressure losses [39].

Even after involving a number of simplifications, the Ho and Li model very closely addresses the physical mechanism of CAF pressure losses. This model allows predicting the pressure gradients using the values of oil and water flow rates. As expected, this two-fluid model underpredicts the CWAF friction losses consistently. This is because a CWAF system involves considerable wall fouling and oil core eccentricity, while the two-fluid model was developed for the perfect CAF in a hydrodynamically smooth pipe.

Adapting the modeling methodology described in [32], a physics-based approach to model CWAF pressure losses was proposed in [38]. Please refer to Ref. [23] for the details of the development. It is a semi-mechanistic two-fluid model, which requires simulating the turbulent flow of annular water on the fouling oil layer in a lubricated pipeline. The turbulence in the water annulus is modeled with the anisotropic ω-RSM model instead of the standard isotropic models. It can capture the effects of the thickness of the wall fouling layer, the equivalent hydrodynamic roughness produced by the viscous oil layer on the pipe wall, and the water holdup. The model was validated using actual CWAF data collected by varying pipe diameter, oil viscosity, water fractions, and flow rates. Compared to existing CFD models, this model is more robust as it not only produces better predictions but also requires significantly fewer computing resources. Although a promising development, the current version of the model involves some simplifications and is difficult to implement.

2. Descriptions of selected models

2.1 CAF model

Arney et al. [11] performed a comprehensive study on the core annular flow in a horizontal pipeline involving both experiments and theoretical analysis. Their primary objective was to enrich the CAF database and introduce a simple approach to calculate the frictional pressure losses.

For the experiments, Arney et al. [11] used two oils: waxy crude oil (ρ = 985 kg/m^3 and μ = 0.6 Pa s) and No. 6 fuel oil (ρ = 989 kg/m^3 and μ = 2.7 Pa s). The experimental setup consisted of three pipeline segments made of a glass pipe having 15.9 mm inner diameter (ID). The 6.35-m long first part was used for flow visualization using a Spin-Physics SP2000 high-speed video system and a 35 mm camera. The second part of the pipeline was used to connect two pressure tap. 1.42 m apart. The last part of the pipe was 1.47 m long and utilized to measure the *in situ* volume fraction of water, that is, water holdup.

Two important parameters used for this study were the water holdup (H_w) and input water fraction of (C_w). It was observed that the H_w was consistently larger than the C_w. That is, the oil core in CAF was moving faster than the annular water phase. Similar experimental finding was also reported in [23]. Areny et al. [11] then collated all the previous CAF experimental data from the literature with their own measurements to propose the following correlation between H_w and C_w:

$$H_w = C_w[1 + 0.35(1 - C_w)] \tag{1}$$

They also measured the pressure losses for a variety of flow conditions. Based on the data, they proposed a single-fluid model. The friction factor (f) was correlated to a system specific Reynolds number (Re_a):

$$f = 0.079/Re_a^{0.25}, Re_a > 4000 \tag{2}$$

$$Re_a = \rho_c DV/\mu_w \tag{3}$$

$$\Delta P/L = f\rho_c V^2/2D \tag{4}$$

where $\Delta P/L$ is the pressure gradient, μ_w is the water viscosity, and ρ_c is an equivalent fluid density. The viscosity of the equivalent liquid was considered to be equal to that of water (μ_w). Empirical expression used to correlate the density of this hypothetical liquid (ρ_c) to the densities of oil (ρ_o) and water (ρ_w) is as follows:

$$\rho_c = H_w\rho_w + (1 - H_w)\rho_o \tag{5}$$

Using this model, it was possible to predict a large number of CAF pressure drop data sets with a reasonable accuracy. The model showed good conformance with friction factor values at high Reynolds number. However, there was significant under prediction when Reynolds number was low. This was due to the fact that at low Reynolds number, the core annular flow was slightly unstable.

2.2 SLF model

Joseph et al. [12] investigated the "Self-Lubrication" phenomenon of Bitumen froth (approximately 60% bitumen, 30% water, and 10% solids by volume), which was extracted using Clark's hot water extraction process from the oil sands of Athabasca. The water in the froth, while transporting through the pipelines, was released due to high shear resulting in a lubricating layer near the wall. This is just another form of CAF where the annular water comes from the mixture itself.

Two different setups were used to experimentally study the phenomenon of self-lubrication. First, a setup of 6 m long 25 mm ID pipe loop was used at the University of Minnesota. The froth was continuously recirculated. The duration of the experiments varied from 3 to 96 hours. The velocities for which pressure gradients were measured ranged from 0.25 to 2.5 m/s. Water volume fractions were kept within 20–40%, and the froth temperature ranged between 35 and 55°C. From the collected pressure gradient data at different flow rates, it was observed that there was a critical velocity range (from 0.5 to 0.7 m/s) below which the self-lubrication was being lost. The longest experimental time in this setup was 96 hours. This test was conducted in an already fouled pipeline. Despite the residual fouling, no further fouling was observed during the experiment. The authors suggested that this may be due to the clay particles (Kaolinite) from the released water protecting the oil core from accumulating to the pipe wall. The researchers also observed that heating the froth to a higher temperature would not necessarily improve the lubrication. This was due to the opposing effects of lowered bitumen viscosity at high temperature and reabsorption of the released water into the core. The other test setup (0.6 m ID and 1000 m long pipeloop) was at Syncrude (Canada), where pilot-scale tests were performed.

Based on the experimental data, a single-fluid model for the SLF of bitumen froth was proposed. In this model, a "Blasius-type" equation was used to correlate the f with a water equivalent Reynolds number (Re_w):

$$f = 0.079K_j/Re_w^{0.25} \tag{6}$$

In Eq. (6), the complex flow behavior of self-lubricated flow is addressed with an empirically determined value of K_j. It was assumed to be a function of temperature only (K_j = 23 when temperature ranges 35–47°C and K_j = 16 when temperature ranges 49–58°C). Water content was considered to have negligible effect on K_j. Frictional pressure losses are 15–40 times greater when predicted using the above model than those for water flowing alone under identical flow conditions. The application of this model for predicting LPF pressure losses is extremely limited according to previous researches [10, 13].

2.3 CWAF model 1

McKibben et al. [13] carried out the investigation to examine free water-crude oil flows and, specifically, to establish a correlation for predicting the pressure gradients in continuous water assisted flow.

The experiments were conducted using the followings:

a. A 53-mm ID pipeline consisting of approximately 60 m long horizontal insulated section;

b. The water fractions between 0.10 and 0.36;

c. The temperatures ranging from 18 to 39°C

d. The average velocities of the mixtures between 0.5 and 1.2 m/s;

e. Four different oils with the viscosities of 91.6, 24.9, 7.1, and 5.8 Pa s.

On the basis of the CWAF data sets at different water equivalent Reynolds number, the correlation of Fanning friction factor was found as:

$$f = \frac{1410}{Re_w} \tag{7}$$

$$f = \frac{\left(-\frac{dP}{dL}\right)D}{2\rho_w V^2} \tag{8}$$

$$Re_w = \frac{DV\rho_w}{\mu_w} \tag{9}$$

The inverse relationship between friction factor and water Reynolds number suggested that friction was controlled by a very thin water layer. A water layer of this type formed the lubrication region surrounding the oil core and provided the lubricating force required to overcome the effect of natural buoyancy.

2.4 CWAF model 2

Rodriguez et al. [15] mainly focused on lab- and pilot-scale experimental measurements. They also took a semi-mechanistic approach to model frictional pressure losses of horizontal core annular flows in pipes having both fouled and clean walls. That is, the model was actually developed for the CWAF systems.

Lab-scale tests were conducted with a 27-mm ID PVC pipe and a crude oil having a viscosity of 0.5 Pa s at 20°C. For the pilot-scale experiments, a steel pipeline (77 mm ID and 274 m length) was used to pump a highly viscous crude oil (36.95 Pa s and 972.1 kg/m^3 at 20°C). A freshwater network was used to control the water injection. A piston pump pumped the water, and its flow rate was adjusted via a calibrated frequency inverter. The water superficial velocity was kept constant at 0.24 m/s, and three oil superficial velocities, 0.80, 1.00, and 1.10 m/s, were tested.

In the experiments, a wavy core of viscous oil was observed, and the annular flow of water was mostly turbulent (Reynolds number for the water flow: $1000 < \mathrm{Re}_2 < 14{,}500$, $\mathrm{Re}_2 = \frac{\rho_2 V_2 D}{\mu_2}$). The proposed model first defined the irreversible hydrodynamic component of the frictional pressure gradient ($\frac{\Delta P}{L}$):

$$\frac{\Delta P}{L} = b\left(\frac{\rho_m VD}{\mu_m}\right)^{-n}\frac{\rho_m V^2}{2D} \tag{10}$$

where D is the pipe ID, V is the mixture velocity, ρ_m is the mixture density, μ_m is the mixture viscosity, and b is an empirical constant. The μ_m was obtained by evaluating the ratio between the wall shear stress in core-annular flow (τ_o) and the wall shear stress if the annular water was flowing alone in the pipe at mixture flow rate (τ_w). Assuming the phases have the same density and use the same power law to express the friction factors in both flows, the shear stress ratio (R_τ) was expressed as:

$$R_\tau = \left|\frac{\tau_o}{\tau_w}\right|_{\rho_0=\rho_2} = \frac{b\left(\frac{\rho_2 V_2 D}{\mu_2}\right)^{-n}\frac{\rho_2 V_2^2}{2}}{b\left(\frac{\rho_2 VD}{\mu_2}\right)^{-n}\frac{\rho_2 V^2}{2}} = \frac{1}{(1-\varepsilon)^n[1+(s-1)\varepsilon]^{2-n}} \tag{11}$$

$$\varepsilon = \frac{1}{1+s\frac{V_2}{V}} \tag{12}$$

where V_2 is the average in-situ water velocity. Also, R_τ is the ratio between the corresponding pressure drops, and from Eq. (11), one obtains

$$R_\tau = \left|\frac{\frac{\Delta P}{L}}{\frac{\Delta P}{L}_{2,o}}\right|_{\rho_1=\rho_2} = \left(\frac{\mu_2}{\mu_m}\right)^{-n} \tag{13}$$

where $\frac{\Delta P}{L}_{2,o}$ is the extrapolated pressure drop for the annulus fluid alone in the pipe at mixture flow rate:

$$\frac{\Delta P}{L}_{2,0} = \lim_{\varepsilon \to 0} \frac{\Delta P}{L} = b\left(\frac{\rho_2 VD}{\mu_2}\right)^{-n} \frac{\rho_2 V^2}{2D} \tag{14}$$

From Eqs. (13) and (14), the mixture viscosity can be expressed as:

$$\mu_m = \frac{\mu_2}{(1-\varepsilon)[1+(s-1)\varepsilon]^{(2-n)/n}} \tag{15}$$

Eq. (15) shows that the mixture viscosity is affected by the slip ratio: the faster the core moves relative to the annulus, the lower the mixture viscosity and pressure drop. For the simple case of perfect core annular flow (PCAF), putting n = 1 and s = 2, Eq. (15) can be transformed into:

$$\frac{1}{\mu_m} = \frac{1-\varepsilon^2}{\mu_2} \tag{16}$$

The final form of the pressure drop model was obtained by introducing a two-phase multiplier defined as

$$\varnothing_{2,o} = \frac{\frac{\Delta P}{L}}{\frac{\Delta P}{L}_{2,0}} \tag{17}$$

Using Eq. (10), (13), and (14), the above equation becomes

$$\varnothing_{2,o} = \left(\frac{\rho_m}{\rho_2}\right)^{1-n} R_\tau \tag{18}$$

which becomes using Eq. (12) and $\rho_m = \varepsilon\rho_1 + (1-\varepsilon)\rho_2$:

$$\varnothing_{2,o} = \left[1 - \left(1 - \frac{\rho_1}{\rho_2}\right)\varepsilon\right]^{1-n} (1-\varepsilon)^{-n}[1+(s-1)\varepsilon]^{n-2} \tag{19}$$

The hydrodynamic component of frictional pressure gradient can be expressed as

$$\frac{\Delta P}{L} = b\left(\frac{\rho_2 JD}{\mu_2}\right)^{-n} \frac{\rho_2 J^2}{2D}\left[1 - \left(1 - \frac{\rho_1}{\rho_2}\right)\varepsilon\right]^{1-n} (1-\varepsilon)^{-n}[1+(s-1)\varepsilon]^{n-2} \tag{20}$$

or

$$\frac{\Delta P}{L} = \varphi Q^{2-n}\left[1 - \left(1 - \frac{\rho_1}{\rho_2}\right)\varepsilon\right]^{1-n} (1-\varepsilon)[1+(s-1)\varepsilon]^{n-2} \tag{21}$$

where $\varphi = \frac{b}{2}\left(\frac{\pi}{4}\right)^{n-2} \rho_2^{1-n}\mu_2^n D^{n-5}$, and Q is the mixture flow rate.

The proposed model can be used to analyze, correlate, and generalize pressure drop data. Along with that, the model allows for the satisfactory representation of different annulus flow regimes, kinematic effects, and wall conditions, including fouling. The model accounts for effects of buoyancy on the core. However, it cannot provide reliable predictions without regressing the values of b and n on the basis of reliable data set.

2.5 CWAF model 3

In continuation of the previous research [13], McKibben et al. [10] carried out an extensive experimental investigation of CWAF. The tests were conducted at the Saskatchewan Research Council (SRC), Saskatoon, Canada using 25, 100, and 260 mm steel pipe flow loops. The average thickness of wall fouling (t_c) was estimated using a special double-pipe heat exchanger [19]. The estimations were validated by using a hot-film probe to measure the physical thickness of the fouling oil layer. Heavy oils having viscosities in the range of 0.62–91.6 Pa s were used for the experiments. The input water fraction was within the range of 30–50%. Additional details of the experimental facilities are available in [23].

Based on the experimental study, a new empirical correlation for the Fanning friction factor was proposed as follows:

$$f(CWAF) = 15\left(V/\sqrt{gD}\right)^{-0.5} f_{CF}^{1.3} f_{OIL}^{0.32}\, C_{CF}^{-1.2} \tag{22}$$

where V is the average mixture velocity, g is the gravitational acceleration, D is the pipe diameter, f_{CF} is the friction factor of aqueous phase, f_{OIL} is the friction factor of oil phase, and C_{CF} is the total volume fraction of water in the mixture. It is a phenomenological model, which is claimed to take into account the effects of inertia, gravity, lubricating water, wall fouling, and viscous oil in CWAF. A large data set comprising more than 300 data points were used for the empirical derivation of the model constants.

2.6 CFD models

A scientific methodology of modeling single phase turbulent flow is to use computational fluid dynamics (CFD) [40]. In general, this modeling approach decomposes the turbulent flow into two parts: (i) time-averaged mean motion; (ii) time-independent fluctuations. The product of such decomposition is the transformation of Navier-Stokes (NS) equations into Reynolds Averaged Navier Stokes (RANS) equations [41]. In course of the mathematical transformation, additional terms of turbulent stresses are produced to make the matrix of equations “unclosed”; that is, the number of unknowns is higher than the number of equations. Various turbulent stresses in RANS equations are necessarily modeled empirically for the “closure” of the matrix [42]. The continuity and RANS equations can be presented with the following simplified differential equations:

$$\frac{\partial U_i}{\partial x_i} = 0 \tag{23}$$

$$\frac{\partial U_i}{\partial t} + U_j \frac{\partial U_i}{\partial x_j} = -\frac{1}{\rho}\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left(\frac{\mu}{\rho}\frac{\partial U_i}{\partial x_j} - \tau_{ij}\right) + S_i \tag{24}$$

where x_i represents the coordinate axes, U_i is the mean velocity, p is the pressure, ρ is the density, μ is the viscosity, S_i is the sum of body forces, and τ_{ij} represents the components of the Reynolds stress tensor. The available models for τ_{ij} can be divided in the categories of eddy-viscosity models and Reynolds stress models [43, 44].

Eddy-viscosity models were developed based on the concept of a hypothetical term known as eddy-viscosity (μ_t), which is considered to produce turbulent stresses caused by macroscopic velocity fluctuations [41]. These models can further be divided into three major groups, namely zero-equation, one-equation, and two-equation models [43, 44]. Two-equation models, instead of zero- and one-equation models, are generally used at present to solve complex engineering problems [43, 45]. The most commonly used two-equation models are the k-ε and k-ω models [44]. A significant limitation of this group of models is that they are meant to describe isotropic turbulence [46, 47]. That is, only the significant components of the Reynolds stresses can be computed with the two-equation models. As a result, the group of two-equation models is practically limited to flows where anisotropy is not important [47, 48]. It should be mentioned that the turbulent water annulus in a CWAF pipeline can experience both anisotropy and rough surfaces [10, 12, 15, 30]. These models are also not suggested for turbulent flow in narrow channels and over very rough surfaces [49, 50].

Anisotropic turbulence can be addressed using Reynolds Stress Models (RSM), in which the hypothetical concept of eddy-viscosity is discarded [46]. An example of anisotropic model is ω Reynolds Stress Model (ω-RSM) [45]. In this model, the closure for Reynolds stresses is obtained by using seven differential transport equations [42]. It is a higher level and more elaborate modeling approach compared to the isotropic two-equation models. This kind of models is more widely applicable compared to eddy-viscosity models [43–46]. However, this flexibility is gained through a high degree of complexity in the computational system. The solution of an increased number of transport equations requires significantly higher computational resources compared to the applications of different two-equation models. Even so, a Reynolds stress model was successfully applied to simulate flow conditions that involve anisotropy and rough surfaces [47–51].

To acknowledge the superiority of a Reynolds stress model, a study of the equivalent hydrodynamic roughness (k_s) produced by a wall fouling/coating layer of viscous oil ($\mu_o \sim 21$ kPa.s) was conducted using a rectangular flow cell [23]. The oil surface became rough when turbulent water ($Re_w > 10^4$) was pumped through the flow cell. The rough viscous surface produced very large values of k_s compared to the similar values produced by a clean surface. The relative performance of k-ω model and ω-RSM is presented in **Figure 3**. The ω-RSM can provide reliable predictions of the measured values of friction losses, while the k-ω model yields significant under predictions. This is because the process conditions involved turbulent flow, a hydrodynamically rough surface, and a narrow flow channel, which produced anisotropic turbulence. Comparable analysis was also conducted involving various rough surfaces like solid wall, sandpapers, wall-biofouling layers, and wall-coating layers of heavy oils in different flow cells. Invariably, the ω-RSM always allowed for reliable predictions, while the k-ω model failed to do so. This analysis along with the supporting literature evidently prove that a RSM would be a better choice than a two-equation model to simulate flow conditions, which involve anisotropy and hydrodynamically rough surface.

It should be mentioned that turbulence is a complex subject. Even though RANS methodology is feasible to computationally resolute the phenomenon of turbulence, it averages the process variables with a steady-state assumption in course of solving NS equations. The minor scale unsteady features of turbulence are usually neglected in this kind of averaging [44]. Most important of these features is the turbulent eddies. The scale of these eddies can vary over orders of magnitude [46]. The CFD solution of taking the effect of these eddies into account is computing differential NS equations without any kind of modeling. The available methods for the purpose are Large Eddy Simulation (LES) and Direct Numeric Simulation (DNS). However, these two simulation techniques demand extremely high computational resources [44]. At a computing rate of 1 gigaflop, the requirement of computational time for DNS is of the order of the Reynolds number to the third power (Re^3). Similar

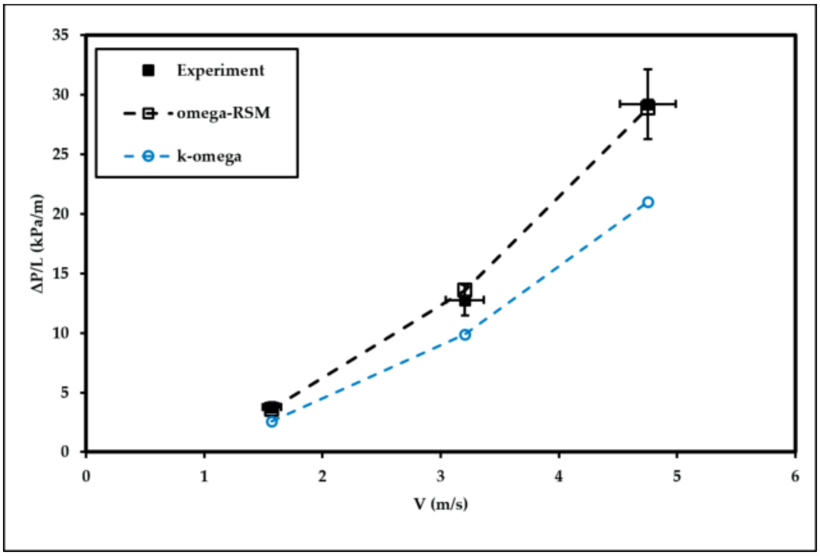

Figure 3.
Comparison of experimental pressure gradients with simulation results: 25.4 × 15.9 × 2000 mm rectangular flow cell; average coating thickness, t_c *= 1.0 mm; equivalent hydrodynamic roughness,* k_s *= 3.5 mm;* $10^4 < Re_w < 10^5$ *[23].*

requirement in LES is generally ten times less than DNS. As an industrial flow system like CWAF pipeline can involve *Re* in the order of 10^5 or higher, application of LES or DNS is not realistic for CWAF at this point.

Two-fluid CFD modeling approach to predict frictional pressure losses of core annular flow was used in different studies [14, 35, 36]. They considered the water annulus to be turbulent and the viscous core to be a laminar plug. Usually, the annular turbulence was modeled with standard k-ε and k-ω models using commercial CFD packages like ANSYS CFX. CFD simulations were also conducted using FLUENT for horizontal oil-water flow with viscosity ratio $\left(\mu_o/_{\mu_w}\right) = 18.8$ in different flow regimes, namely core annular flow, oil plugs/bubbles in water, and dispersed flow [37]. In FLUENT, the volume of fluid (VOF) model of multiphase flow and the SST k-ω scheme of turbulence closure was applied to simulate the oil-water flow. The SST k-ω turbulence scheme at the interface provided better predictions than the standard k-ε and re-normalization group (RNG) k-ε models. Although these turbulence models might show some superiority over Prandtl's mixing length model used in [32], they are meant for isotropic turbulence and are not suggested for the turbulent flow that involves anisotropy or very rough surfaces [49–51]. In addition, this modeling approach is also expensive computationally as it requires solving the governing equations for both phases of oil and water. Using an anisotropic model makes this modeling approach even more expensive from a computational perspective. Moreover, the interphase transfer of mass and momentum is modeled in this methodology by using the default mixture model ANSYS CFX or FLUENT. The correlations used for these models are not validated for the interfacial mixing of LPF systems.

3. Comparative analysis

The performance of existing models in predicting friction losses is analyzed by comparing the experimental results collected for a lab-scale LPF system with the corresponding results obtained using five different models in **Figure 4**. The water equivalent friction factor (f_w) and Reynolds number (Re_w) defined as follows are presented in this figure:

$$f_w = \frac{\Delta P}{L} \frac{D}{2\rho_w V^2} \tag{25}$$

$$Re_w = \frac{D U_w \rho_w}{\mu_w} \tag{26}$$

where $\Delta P/L$ is the pressure gradient, D is the internal pipe diameter, V is the bulk velocity, U_w is the water superficial velocity, and ρ_w and μ_w represent water density and viscosity, respectively. The experiments were conducted in a 26-mm horizontal PVC pipeline to collect data under typical CWAF operating conditions [7, 34]. The heavy oils used for the experiments had densities and viscosities in the ranges of 900–950 kg/m^3 and 3.3–16.0 Pa s, respectively. Although different flow patterns were observed in course of the experiments, the core annular flow with oil fouling on pipe wall, that is, the CWAF was the dominant regime under a wide range of flow conditions. The results shown in **Figure 4** reveal the major limitation of the existing models to be their system specificity.

The CAF model proposed by Arney et al. [11] significantly under predicts the experimental results for CWAF tests. The model was developed based on the experiments conducted in a 15.9-mm glass pipeline, which was selected for the purpose of controlling wall fouling and visualizing the flow regime. Compared to the CAF model, higher predictions of f_w-values by the SLF model and CWAF model 2 can be seen in **Figure 4**. These three models were developed for CAF systems having different degrees of wall fouling and intermittent lubrication. It is interesting to note that the trends of the results produced by all of these three models are similar to that of Blasius law predictions, which represent the frictional losses as only water flows through a pipe. This is because these models were developed based on the Blasius correlation. Clearly, the methodology of modeling CWAF friction losses by modifying Blasius law is not a successful approach. On the other hand, the CWAF model 1 also fails to provide satisfactory predictions. It was developed by modifying the standard f correlation for laminar flow regime ($f = 64/Re$).

Among the models presented in **Figure 4**, the CWAF model 3 demonstrates superiority in predicting both the magnitude and the trend of CWAF friction factors. The performance of the model tends to be better at higher flow rates. The reason is, most likely, the difficulty in establishing the lubricated flow regime at lower flow rates. The improved predictions of this model can be attributed to the following facts:

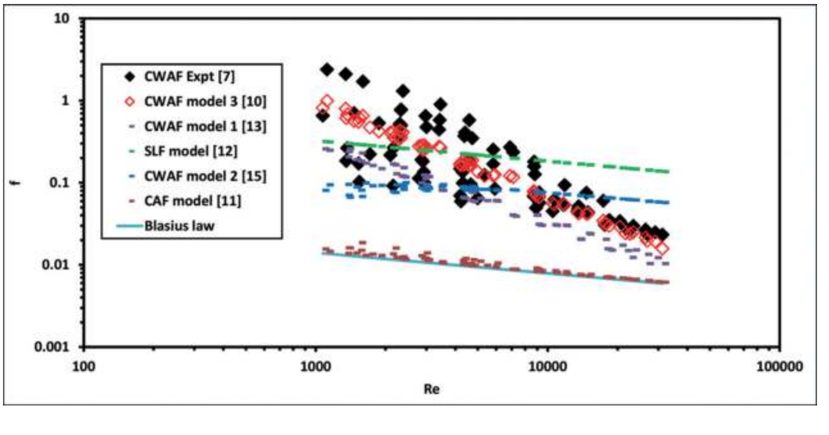

Figure 4.
Comparison of experimental and predicted values of water equivalent friction factor with respect to an equivalent Reynolds number.

i. Instead of a purely empirical reasoning, the model was developed based on phenomenological observations. The physics of frictional pressure losses in a CWAF pipeline dominated the modeling approach. It can be considered as an empirical two-fluid model.

ii. Special attention was paid to the contribution of wall fouling in increasing CWAF pressure drops.

iii. A large data set covering a wide variety of process conditions was used to regress out the model constants.

However, it should be emphasized that the CWAF model 3 needs further validation with multiple independent data sets. Producing more quality data of CWAF pressure losses would be essential for the purpose.

As demonstrated here, the existing models generally adopt an empirical approach to predict LPF friction losses. The effects of operating conditions in these models are usually accounted for with empirical constants. The actual physical mechanisms that govern the pressure losses in an LPF pipeline are almost entirely disregarded. As a result, the models developed for a CAF system cannot be applied to a CWAF system and vice versa. More rigorous studies are essential to develop a fully mechanistic approach to model the LPF friction losses. The CFD can be expected to play a significant role in the process.

4. Challenges and opportunities

The lab-scale applications reported in the literature and a few commercial successes prove that continuous water assisted flow is a reliable method for the long-distance transportation of heavy oil. One of the major barriers to spread commercial applications of this flow technology is the lack of a viable model to predict frictional losses on the basis of operating parameters, such as pipe diameter, flow rates, fluid properties, and water fraction. A new model capable of dealing with the hydrodynamic effects produced by the "wall fouling" layer in a CWAF pipeline is required to facilitate wider industrial implementation of this pipeline transportation technology. The phenomenon of wall fouling has not been thoroughly probed till now, although it is an important characteristic of CWAF technology. Therefore, the future researches should be focused on the investigation of the hydrodynamic effects produced by the wall fouling layer.

The oil core touching the pipe wall in a large water-assisted pipeline is another unaddressed phenomenon. Experiments conducted at Saskatchewan Research Council suggest that this phenomenon is significant for intermittent water assist when the bulk velocity is less than 1 m/s and the water fraction is less than 30% [10]. More devoted research works are necessary to determine the contribution of intermittent core/wall contact to the LPF pressure loss.

The presence of solids like sands or clays in a CWAF pipeline is another important issue. The solids embedded on the wall fouling layer and the oil core may increase the equivalent roughness. In some cases, the fine particles in the lubricating water can change its apparent viscosity and the nature of contact between the oil-covered wall and the oil core (see [12] for additional details). Future work in this field would help to characterize the effects of solid fraction on the pressure losses in CWAF pipelines.

5. Conclusions

The objective of the current chapter is to provide a brief introduction to the water lubricated transportation of heavy oil. The contents are summarized as follows:

i. Lubricated pipe flow is an alternative technology for long distance transportation of heavy oil. This kind of water assisted pipeline transportation is more economic and environmentally friendly than the conventional viscous oil transportation technologies. Its applications can be categorized as core annular flow, self-lubricated flow, and continuous water assisted flow. From an engineering perspective, CWAF is more significant than CAF or SLF.

ii. A technical challenge to the field-scale application of CWAF is the absence of a reliable model to predict friction losses.

iii. The models proposed to date for CWAF friction losses can be categorized as single-fluid and two-fluid models. In general, the methodology followed to develop a two-fluid model is more mechanistic, while a single-fluid model is an empirical development.

iv. Applicability of an existing model for a specific set of flow conditions cannot be justified at present without a comparative analysis based on a reliable data set.

v. The most important research opportunities to develop a more reliable model for CWAF friction losses are related to the following subjects:

    a. Wall fouling

    b. Intermittent water lubrication

    c. Solid laden CWAF

    d. CFD modeling

    e. Enrichment of the CWAF database

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Conflict of interest

The authors certify that they have no affiliations with or involvement in any organization or entity with any financial interest (such as honoraria, educational grants, participation in speakers' bureaus, membership, employment, consultancies, stock ownership, or other equity interest, and expert testimony or patent-licensing arrangements), or nonfinancial interest (such as personal or professional relationships, affiliations, knowledge or beliefs) in the subject matter or materials discussed in this manuscript.

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A part of the manuscript is adapted from the corresponding author's PhD dissertation.

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